TPS
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TEXAS A&M UNIVERSITY
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TEXAS A&M ENGINEERING EXPERIMENT STATION

Vibration and NPSHr ($NPSH_3$), improvement of BB1 two stage pump reliability.

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Baker Hughes

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Abstract

This case study discusses the impeller redesign of 10 **8x15 BB1 two stages** Amine pump in a Gas Sweeting Plant in Malaysia. The original pump was supplied in 1996 and after some design changes it had an history of recurring high vibration and impeller damages such as a cavitation erosion on inlet suction side and disk break at impeller outlet.

Pump performance analysis revealed a service operative point close to Minimum Continuous Stable Flow showing vibration peak both on synchronous frequency and VPF (vane passing frequency) typical of secondary flow effect. After a FMEA analysis it was decided to redesign the impeller.

CFD simulation and Static structural with pressure load profiles have been performed and a model test bench has been set up to compare new and old impeller design focusing on $NPSH_r$ curve and on interaction forces between impeller blades and volute lip. Currently the 10 pumps are operating without showing any vibration problem.

Problem Statement

General Info:

- Two trains 5 and 6 have been commissioned, each having 4 steam turbine driven and 1 motor driven amine charge pump.
- The steam turbine driven pumps have a HPRT (Hydraulic Power Recovery Turbine). An overhung clutch permits operation of pump without HPRT.

Issue on Pump

- High Vibration & Noise
- Cavitation Erosion on Fist Stage Impeller
- Impeller Material Detachment

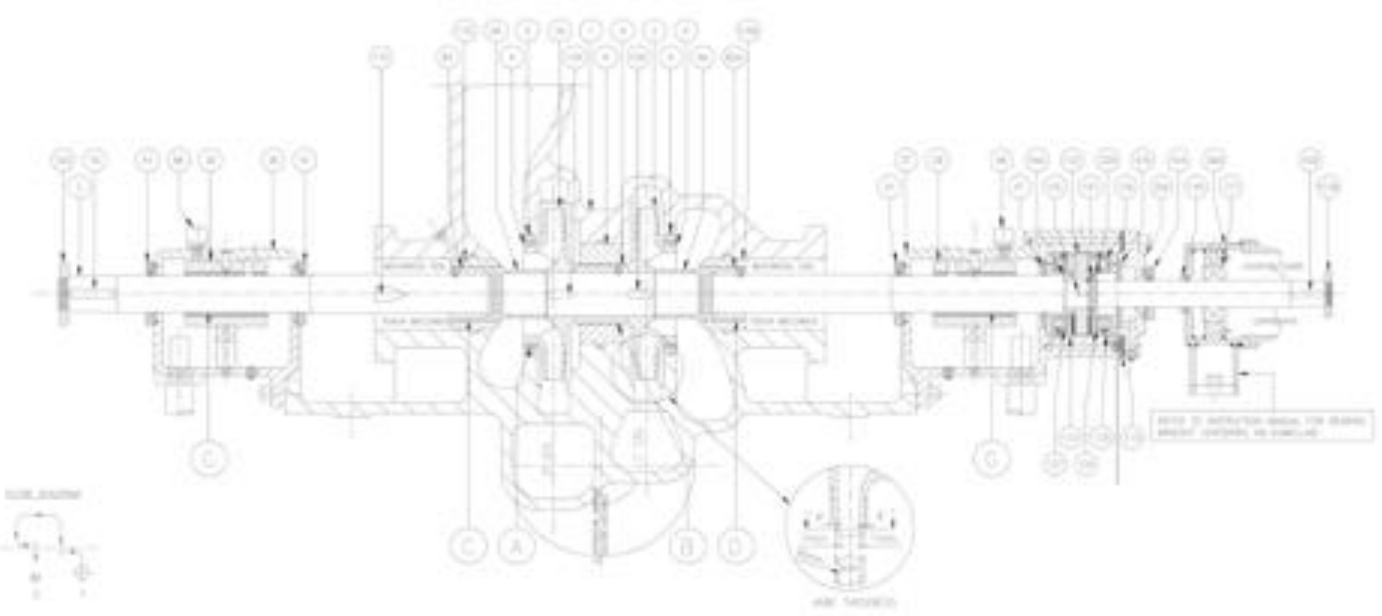


Figure 1 - Impeller Erosion at Suction Indication of Suction Recirculation

Figure 2 - Cracks on Shroud Outer Disk

Problem Statement

- More than 50% of the pump trip and shutdown between 2011-2014 are associated to vibration issue with 6 stop
- 1X - 6X dominant frequency

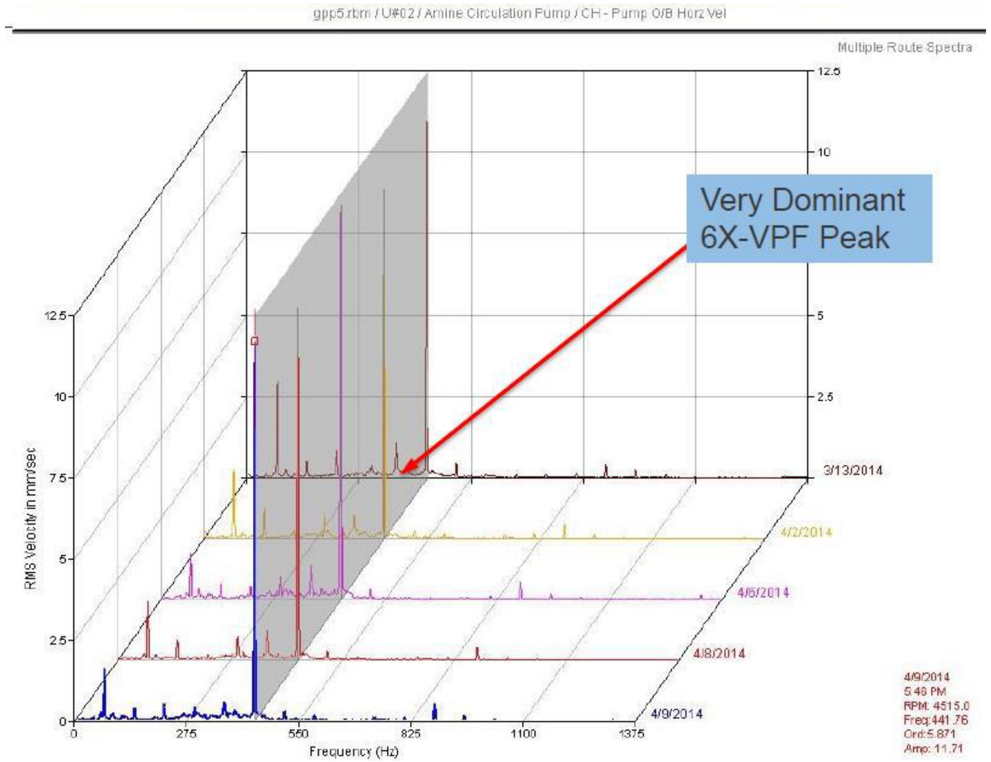


Figure 3 – Waterfall Pump NDE Shaft

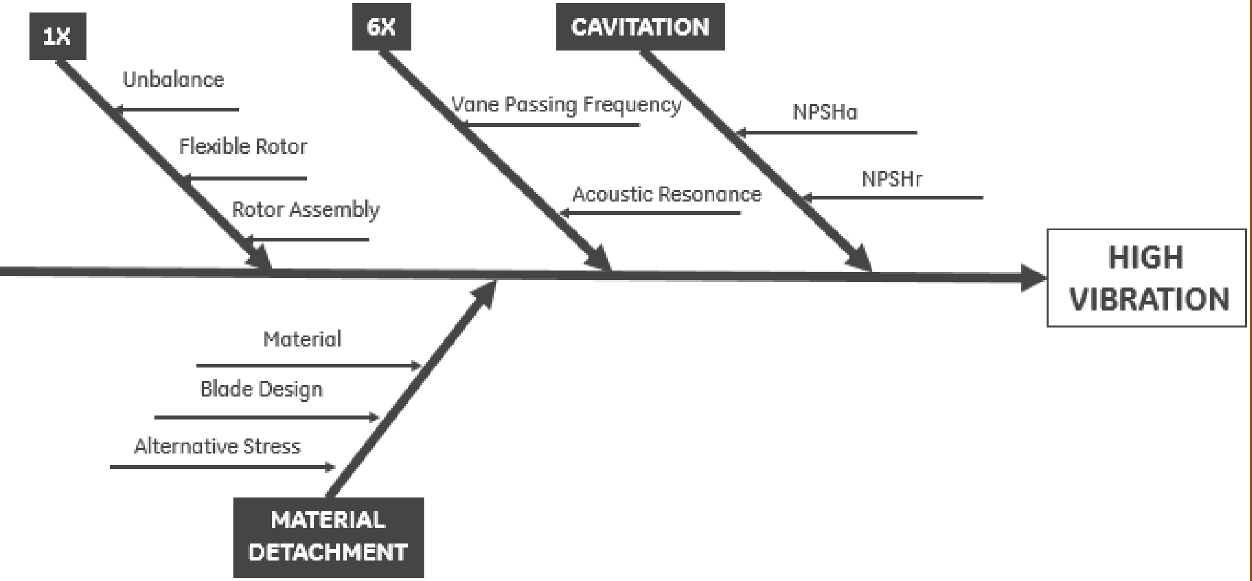
Amine Pump Healthiness	RUN	SEAL POT		VIBRATION						TEMPERATURE				
		DE	NDE	NDE-X	NDE-Y	DE-X	DE-Y	AXIAL-1	AXIAL-2	T-BRG TI-2009	T-BRG TI-2010	NDE-J TI-2011	CASING TI-2012	DE-J TI-2013
	Limit	LL: 20 HH: 100	LL: 20 HH: 100	H: 60 HH: 70	H: 60 HH: 70	H: 60 HH: 70	H: 60 HH: 70	H: 0.6 HH: 0.8	H: 0.6 HH: 0.8	H: 85 HH: 100	H: 85 HH: 100	H: 85 HH: 100	H: 100 HH: 120	H: 85 HH: 100
P5-0201A	-17	8	-1	0.0	0.0	0.0	0.0	1.0	0.2	0.0	0.0	0.0	0.0	34.0
P5-0201B	4432	70	67	46.0	26.5	86.0	88.9	0.3	0.3	73.0	67.0	60.0	55.0	0.0
P5-0201C	0	61	57	1.9	1.8	0.6	0.6	0.1	0.1	36.0	36.0	34.0	36.0	33.0
P5-0201D	4574	59	61	27.5	15.6	32.8	37.8	0.5	0.4	75.0	63.0	73.0	81.0	79.0
P5-0201E	957	64	63	62.4	0.0	61.8	79.0	0.3	0.4	74.0	70.0	72.0	0.0	77.0
P6-0201A	4442	64	67	21.3	20.0	42.2	34.8	0.3	0.2	76.0	62.0	60.0	0.0	81.0
P6-0201B	4498	75	68	92.2	40.0	26.0	25.0	0.7	0.8	89.0	78.0	64.0	83.0	81.0
P6-0201C	-7	62	64	0.8	0.8	0.6	0.7	0.4	0.2	39.0	39.0	43.0	45.0	76.0
P6-0201D	-10	15	-1	0.0	0.0	0.0	0.0	1.0	1.0	0.0	0.0	0.0	0.0	35.0
P6-0201E	1101	55	60	28.5	28.8	21.0	15.8	0.4	0.4	68.0	59.0	66.0	65.0	80.0

Figure 4– Overall No Contact Vibration Value

	DE HOR			DE VERT		
	Overall	VPF	% VPF	Overall	VPF	% VPF
P5-0201A	13.32	12.66	95.0%	3.882	1.845	47.5%
P5-0201B	13.32	12.77	95.9%	3.574	2.227	62.3%
P5-0201C	11.12	9.746	87.6%	2.683	1.626	60.6%
P5-0201D	4.8	0.544	11.3%	4.022	1.693	42.1%
P5-0201E	3.611	1.224	33.9%	2.656	1.411	53.1%
P6-0201A	8.383	8.053	96.1%	2.652	1.331	50.2%
P6-0201B	9.373	7.392	78.9%	3.444	3.332	96.7%
P6-0201C	7.121	7.03	98.7%	3.712	2.744	73.9%
P6-0201D	9.644	9.038	93.7%	2.737	2.133	77.9%
P6-0201E	4.245	1.556	36.7%	3.995	3.42	85.6%

Figure 5– Overall Contact Vibration Value

Fishbone Analysis – Potential Causes and Actions



CAUSES		Synchronous Vibration	Vane Passing Freq.	NPSHr	NPSHa	Material Detachment	Rotor Assembly
ACTIONS	VIBRATION ANALYSIS	X	X				
	RUN OUT REPORT	X					
	MODAL ANALYSIS	X	X			X	
	BLADE DESIGN		X	X		X	
	B-GAP DESIGN		X				
	ACOUSTIC RESONANCE	X	X				
	CFX ANALYSIS			X		X	
	PLANT DATA				X		
	STRESS ANALYSIS					X	
	ASSEMBLY REPORT						X

1X & 6X: Synchronous Vibration and VPF Vibration

Site Vibration and Performance Analysis

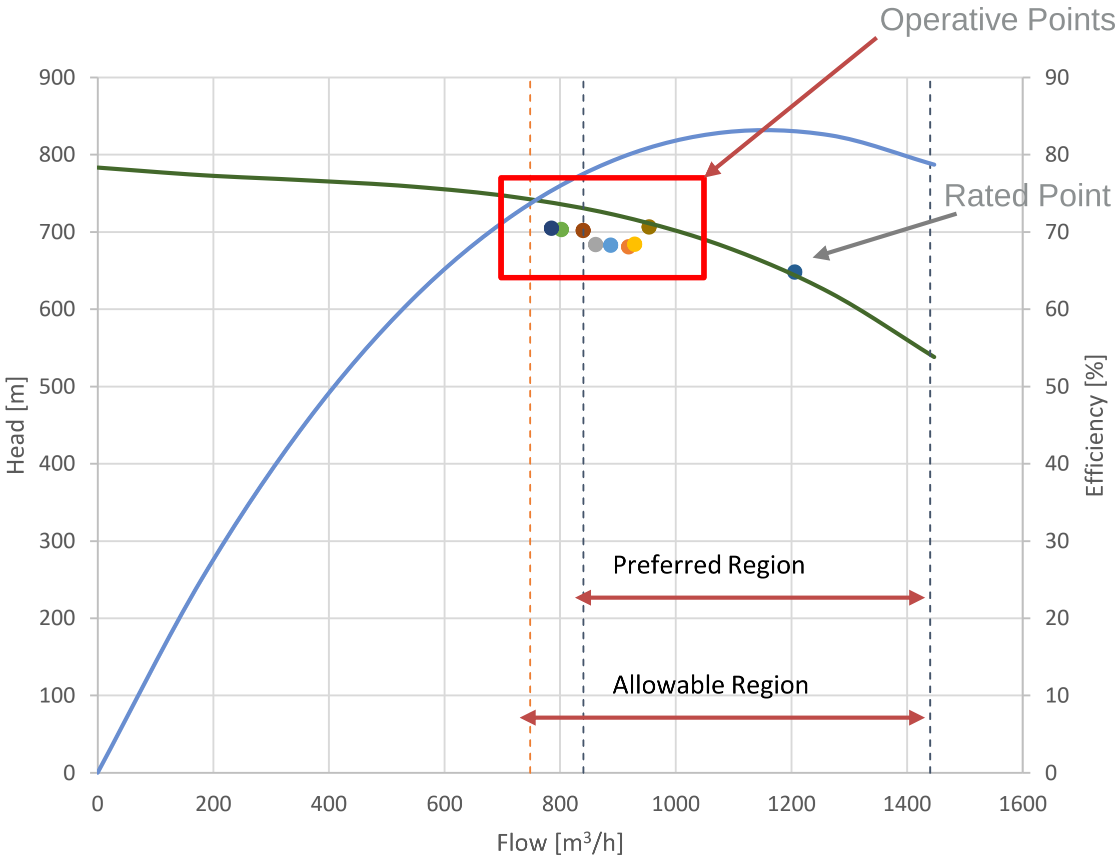


Figure 6 – Site Performance Analysis

- Pump service point range between 0.7-0.8 Q_{BEP} close to Minimum Continuous Stable Flow (MCSF)
- Erosion patterns on inlet blade can be related to recirculation flows due to partial load service
- Recirculation flows induce high vibration at Vane Passing Frequency (VPF) highlighted in spectrum analysis
- A new tailored impeller seized to service point leads to efficiency improvement and a drastic reduction of secondary flows

B & A GAP

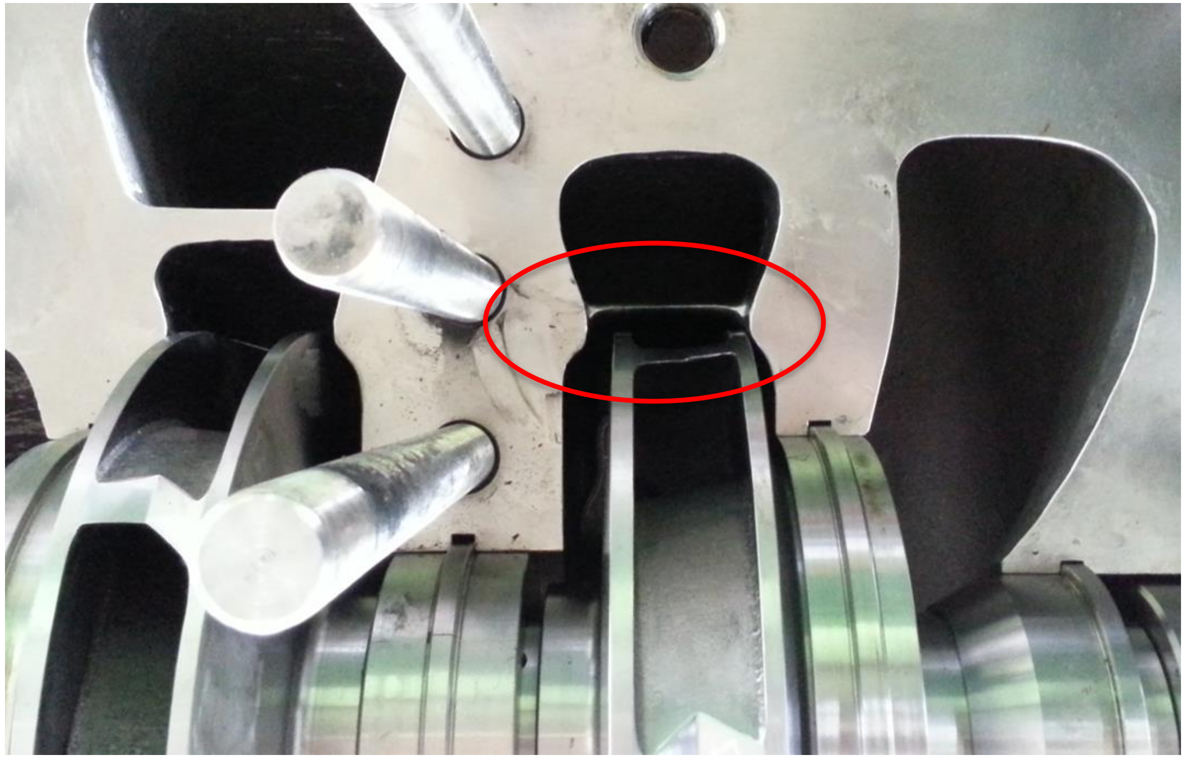


Figure 7 – Pump Rotor Inspection

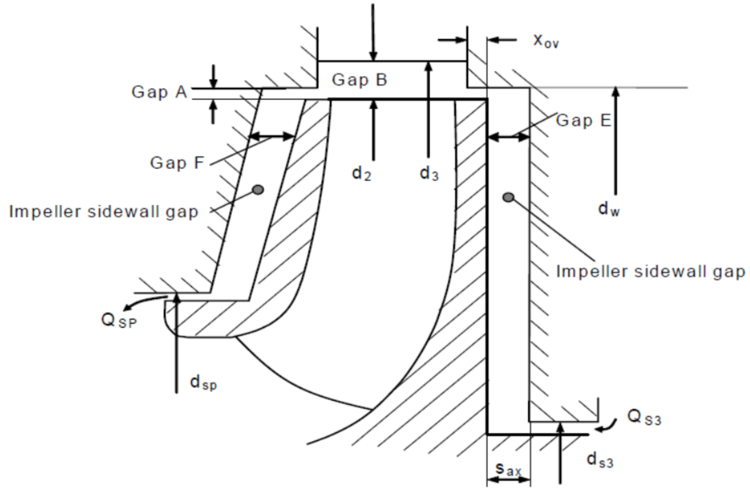


Figure 8– A and B GAP (*Centrifgual Pumps*, Second Ed., J.F. Gulich 2010)

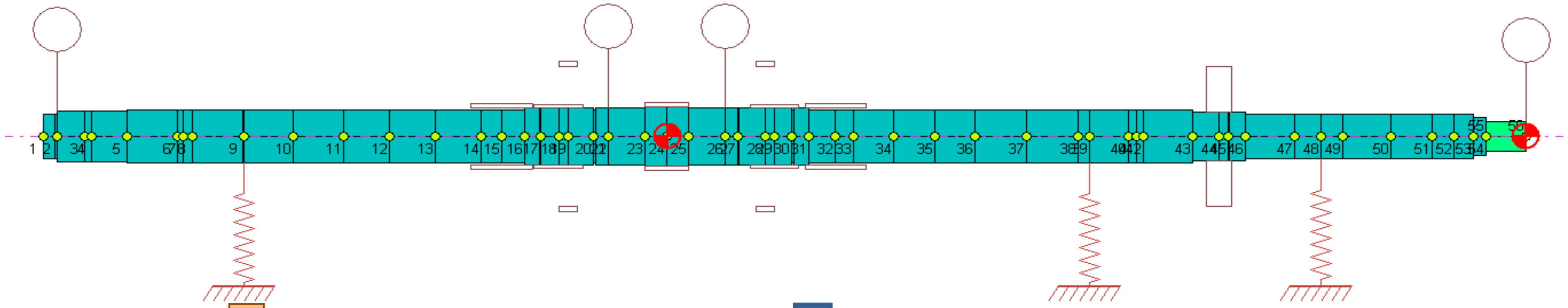
Analysis

- B-Gap lower than API requirement.
- Axial gap out of control

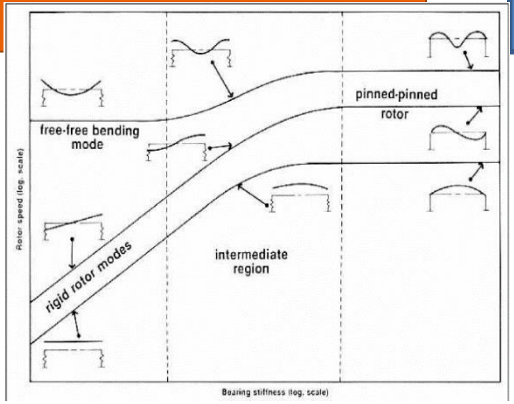
Action

- Reduce impeller outer diameter
- Update assembly procedure for rotor centering and axial clearance check

Rotordynamic Design: Actual Configuration1X



- High shaft velocity
- High bearing sleeve stiffness K
- Low damped bearing C
- Slender shaft



- Low damped pump
- #3 annular sleeves

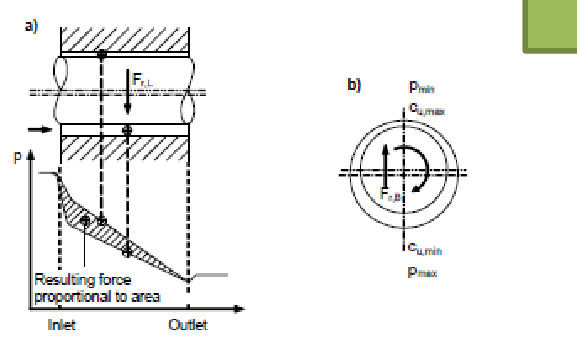
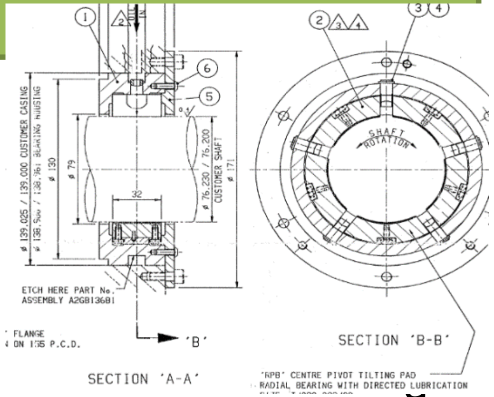


Fig. 10.9. Radial forces in annular seal, a Effect of axial flow (Lomakin effect), b Effect of circumferential flow (Bernoulli effect)

- Suspended clutch
- Hyperstatic configuration
- Slender cantilever beam



Rotodynamic Analysis: Results

- Original pump has been designed according API 610 VII ed with damped unbalance analysis
- Rotor Damped Critical Speeds presents a good separation margin with synchronous excitation
- Low Damped Pump
- Critical 0.5 X and 1X Excited Modes
- External outboard masses (clutch) and Annular Wear Rings impact the vibration response
- Introducing non metallic wear rings will be an increment of the damping factor at high speed

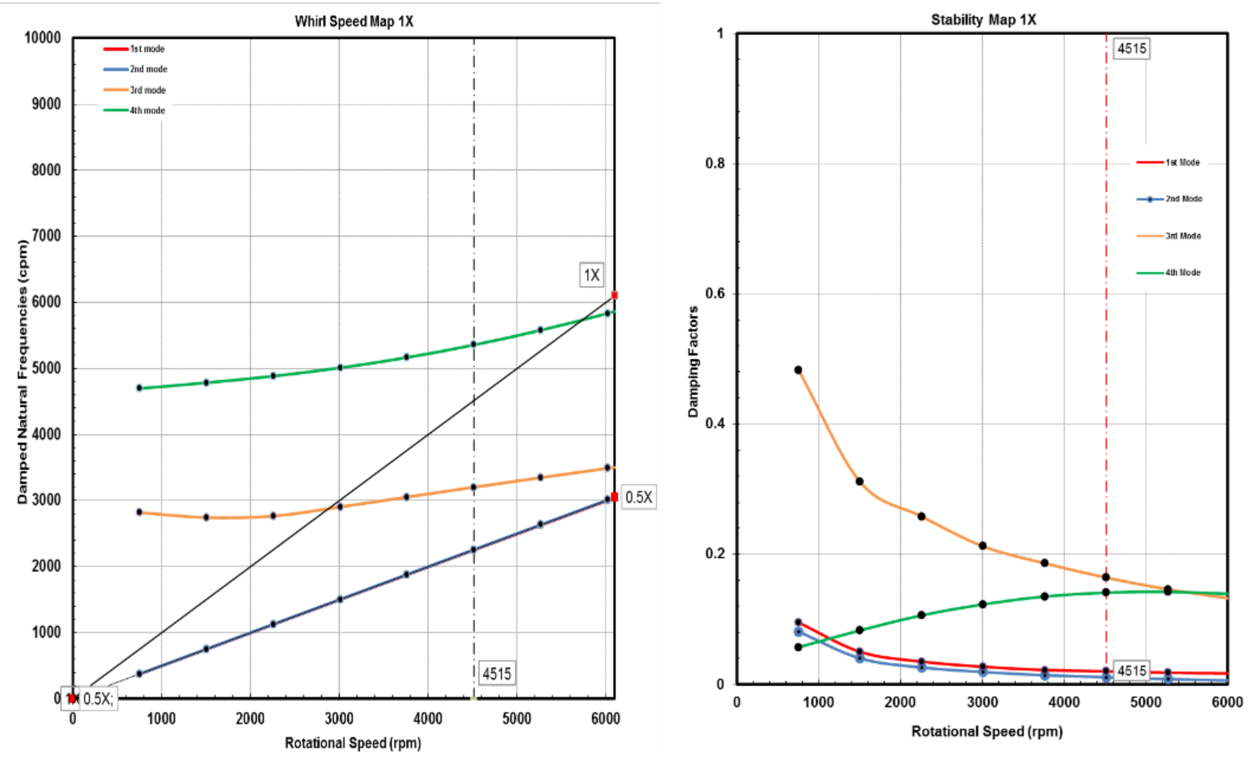
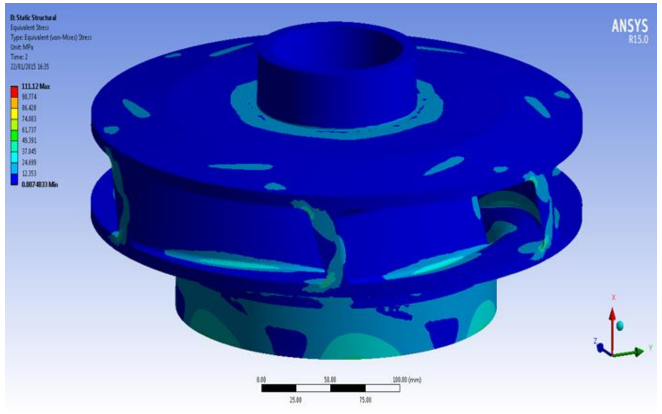
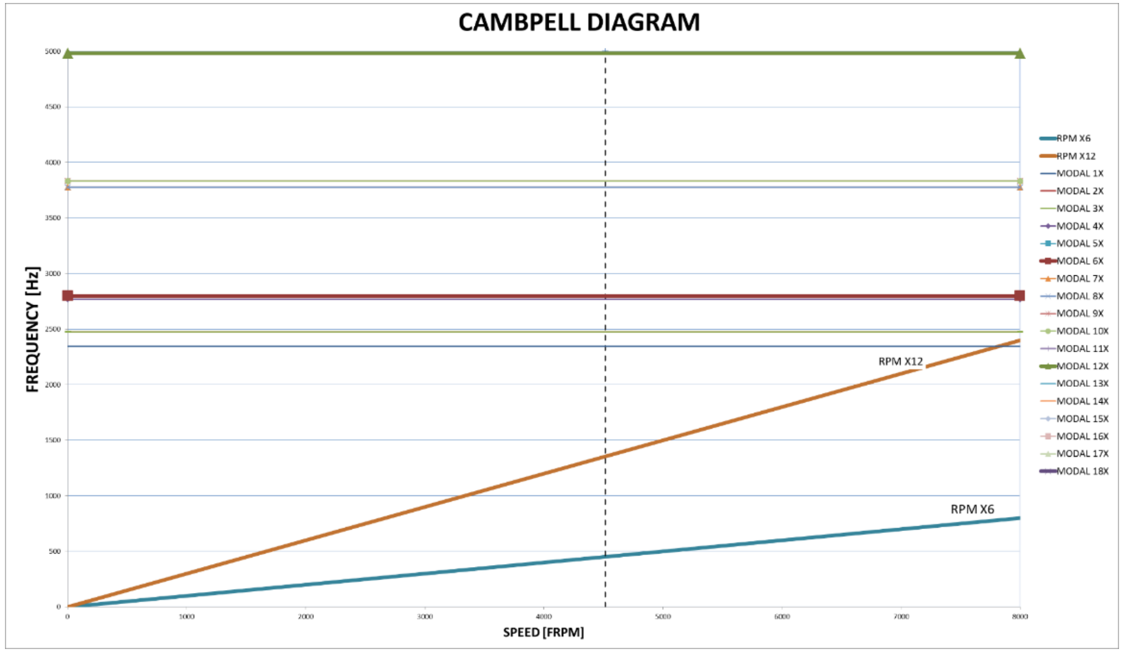


Figure 9– Whirl Speed and Stability Map

Static Stress Analysis and Impeller Modal Analysis





LOADS:

1. Internal pressure Contour
2. Rotational Velocity (4515 rpm)
3. External pressure on impeller disks

Static Stress level **acceptable** (**No** information about **alternative loads**)

Curves intersect at high speed.

Cambpell diagram shows : rotating speed line, harmonic frequencies depending on number of vanes (6X and 12X) and impeller natural frequencies. Intersection point indicates a possible resonance phenomena but a speed far from operating speed.

Bearing Housing Resonance Test

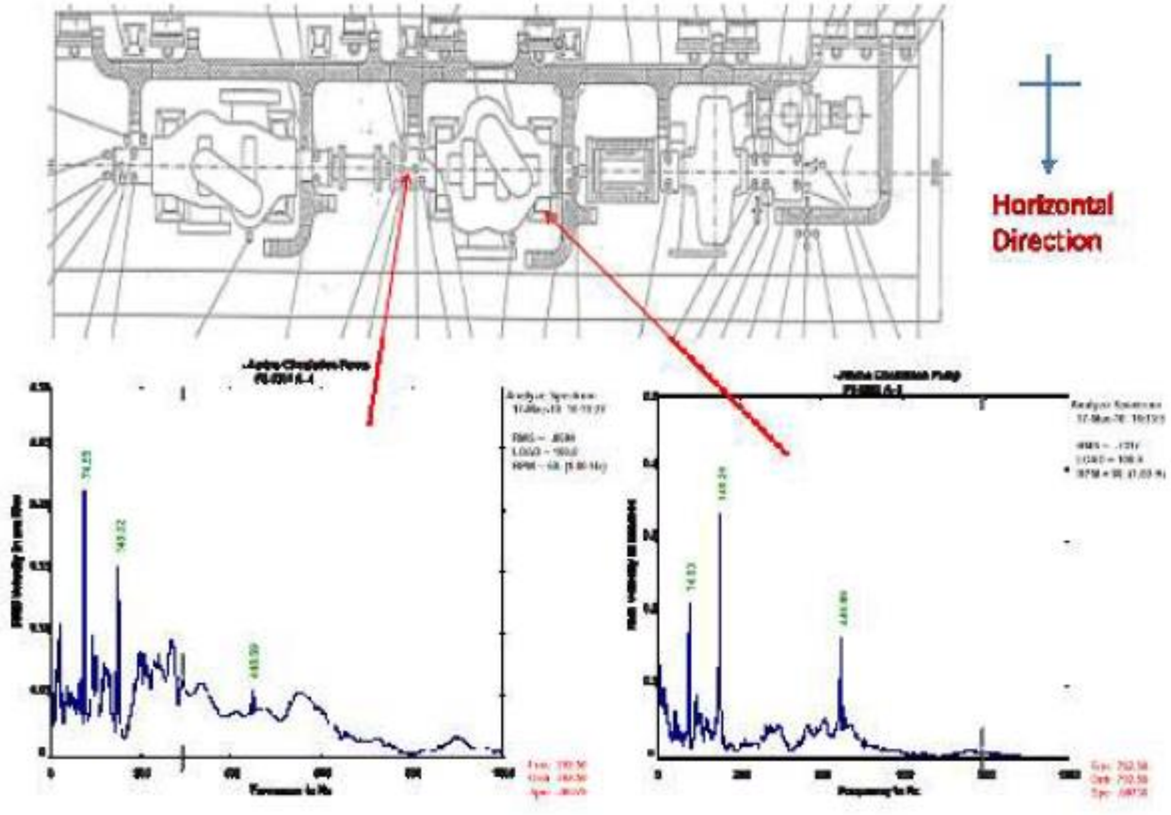
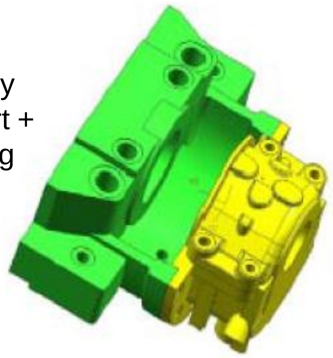


Figure 10 – Bump Test on Site

Due to the absence of 3D model a dummy bearing support + bearing housing has been simulated



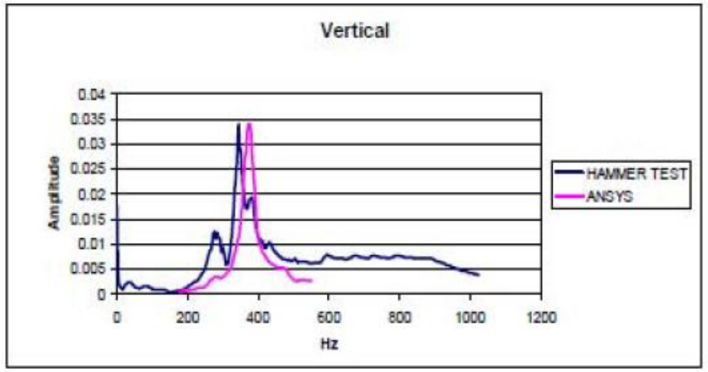


Figure 11 – Ansys and Hummer Test on DE Support

- The bump test indicates a natural frequency of about 446 [Hz].
- VPF (Vane Passing Frequency) is close to 451.5 [Hz]. Vibration on site are excited also by the natural frequency of the bearing support.
- Using a #7 blade impeller design the VPF arises to 527 [Hz]. Lower excitation of bearing support at VPF

Drop Simulation

Single Impeller Multiphase CFX Simulation with Cavitation Model over range of Inlet pressures @ max operating point

Simulation Process fluid : Water
Process Fluid : Amine

NPSH a = 52 m (as per TDS)
NPSH r ≅ 36 m

NPSH margin of about 40%

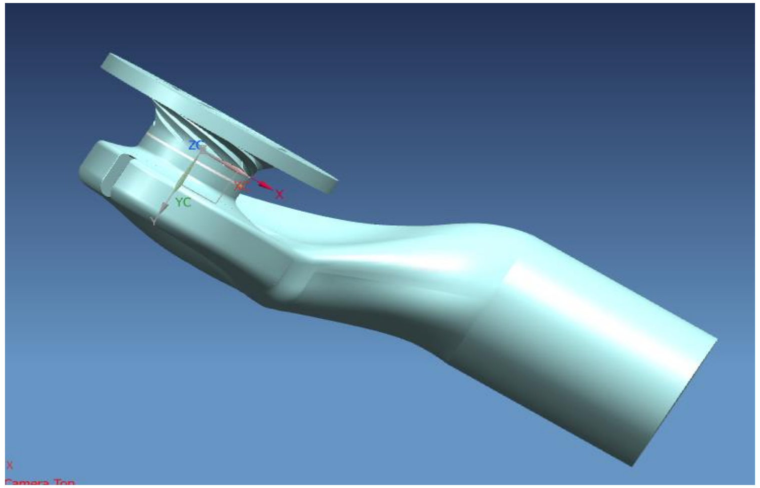


Figure 12 – Simulation Domains

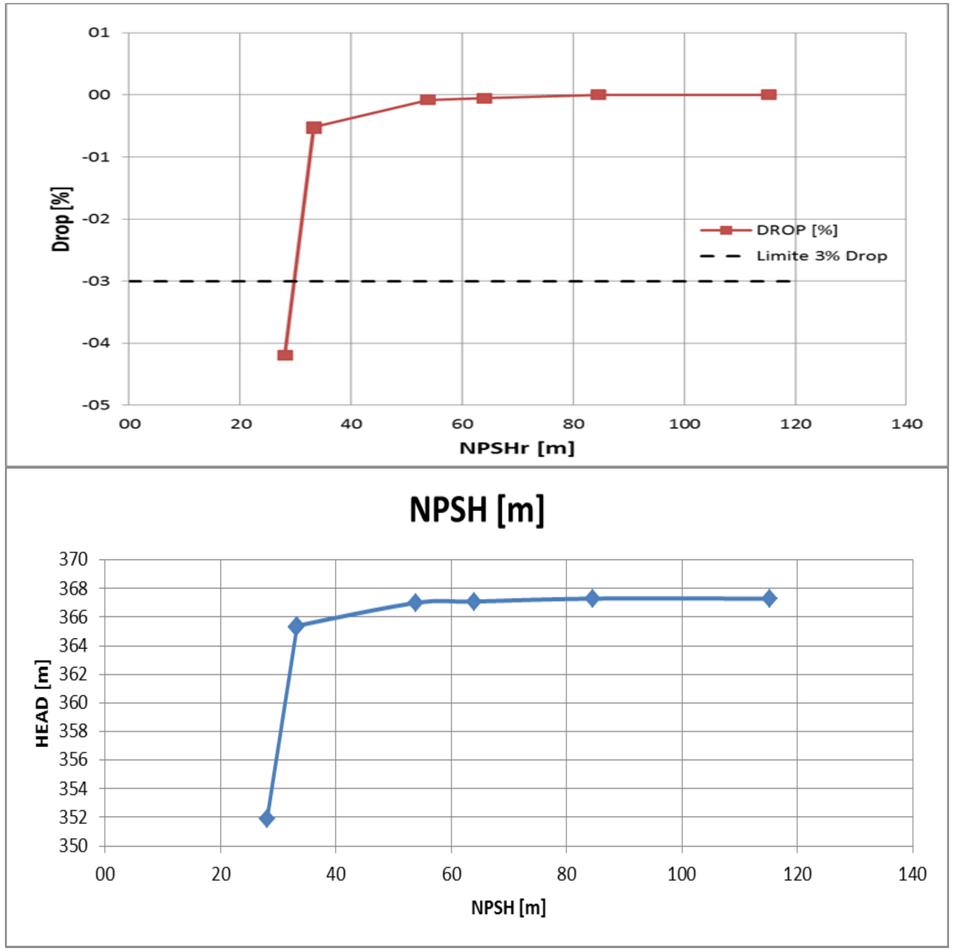


Figure 13 – Drop Cruves

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CFX Analysis of Actual Impeller at D_2 max

Model Details:

- D_2 Max =355,5 mm
- D_2 Cut = 349 mm
- D_3 = 368,5 mm
- Preswirl = 10°

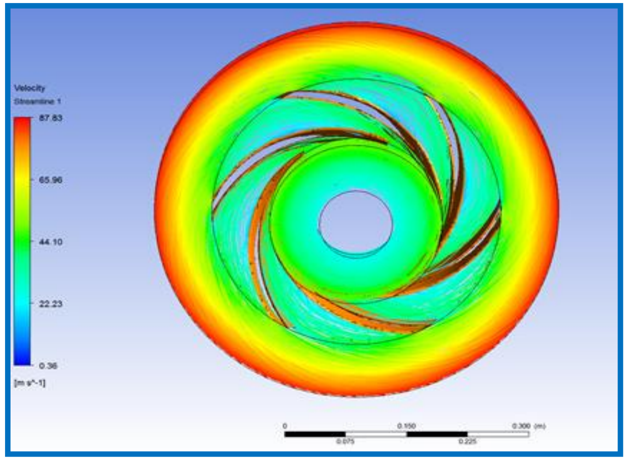


Figure 14 – Impeller domain CFD simulation

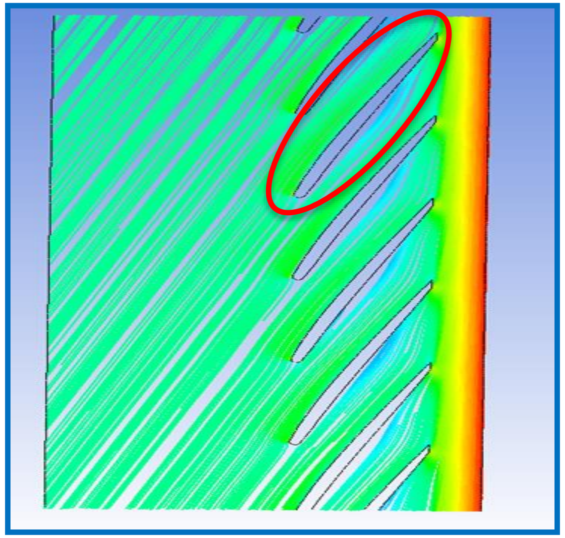


Figure 15 – Impeller streamline

- Validation process shows an high blade load
- This high blade load influence on 6X vibration

Project Execution

- New 7 blade impeller design
- Model test for performance analysis and vibration measurement

Impeller Design Criteria

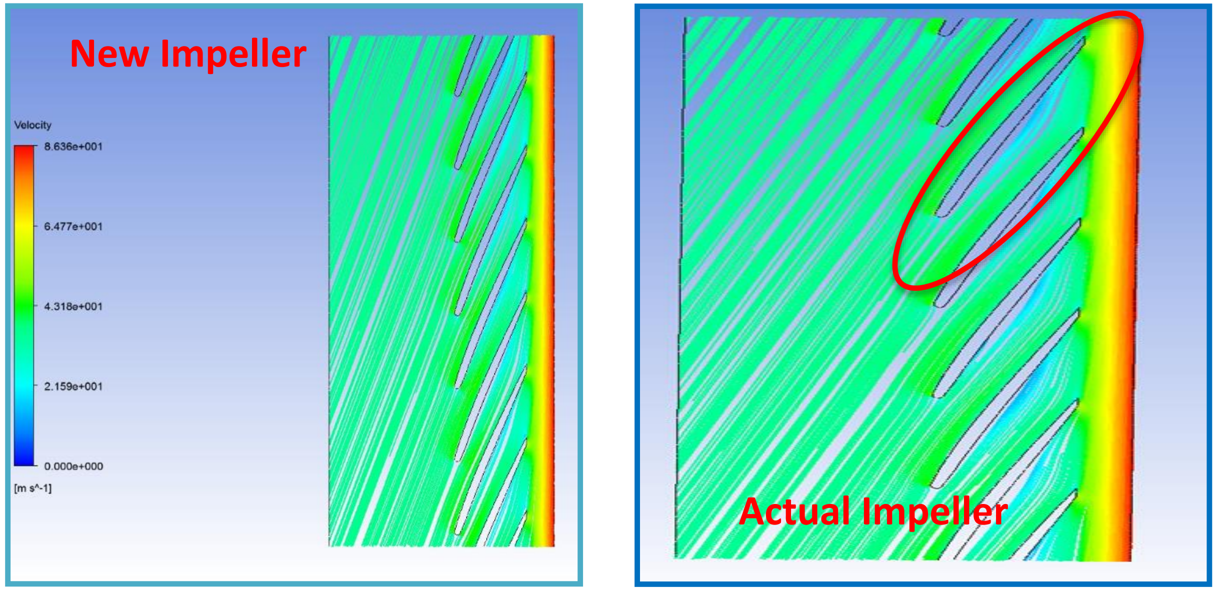


Figure 16– Impeller simulation at max operating point at D2 max

	Actual Impeller	Proposed Impeller
# Blade	6	7
D_2 $[mm]$	355.5	≈ 352
B2 [mm]	39,5	33
Deye [mm]	212	212
CL [-] Blade Loading	1.445	1.281
B_{GAP}	3,65 %	4,69 %
NPSHr [m]	≈ 36	≈ 38

New impeller has been designed to :

- Guarantee the same NPSHr: new impeller have been designed with correct inlet angles at the operating flow (unsteady simulation with suction and impeller domains to define the pre-swirl along meridional radius)
- Reduce recirculation at inlet due to partial load operation (confirmed by customer→ operative points ~64% and 83% of impeller design point)
- Increase the B-GAP and reduce the blade loading to avoid streamline detachment
- Reduce the alternating forces at impeller exit (lean angle)

Model Test Set-Up & Results

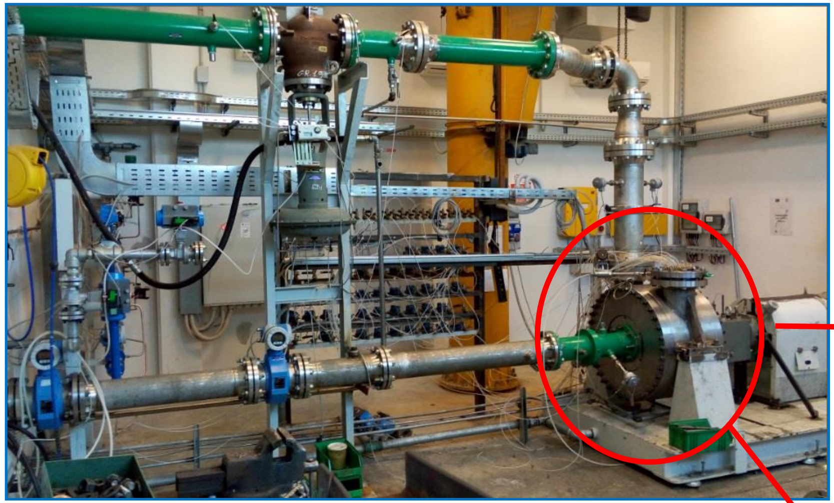


Figure 17 – Test Bench and Flow Loop

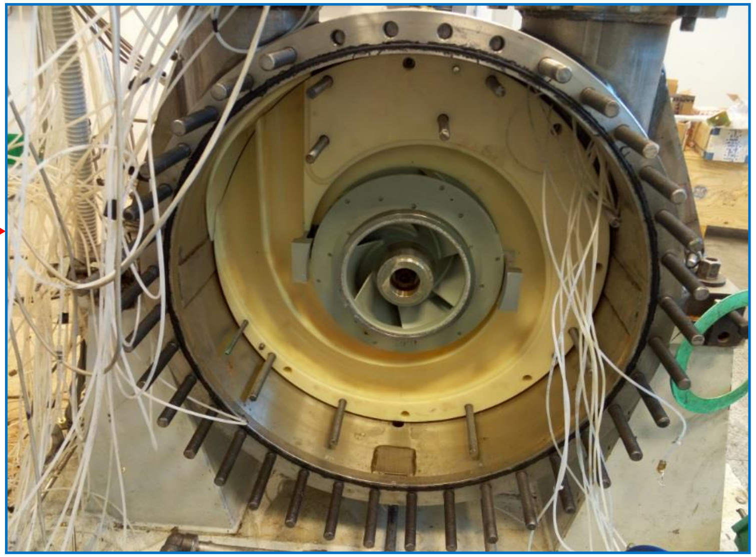


Figure 18 – PrototypeWith Impeller and Discharge Volute Assembled

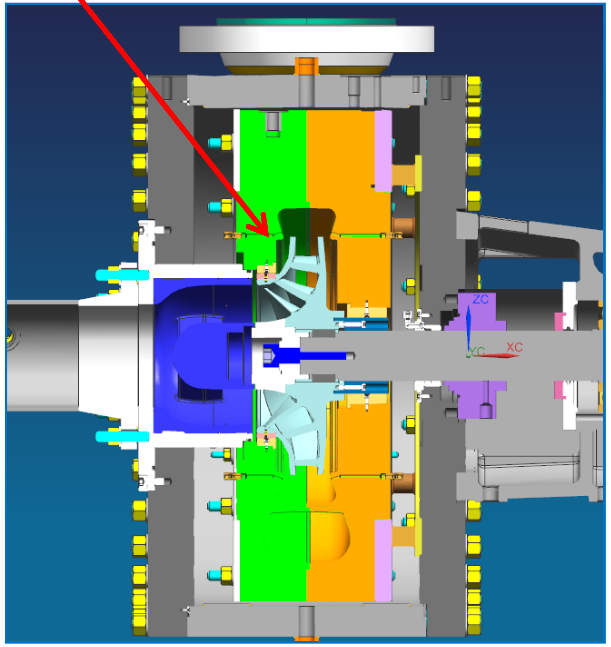


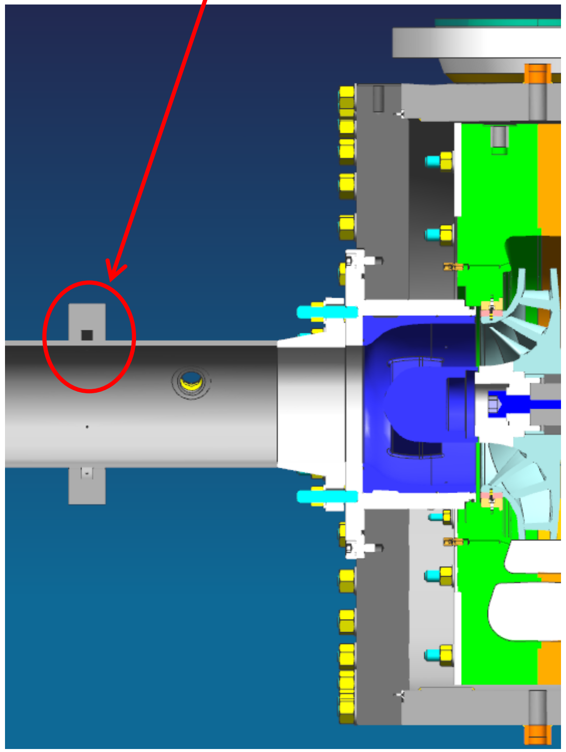
Figure 19 – 3D Prototype Configuration: Single Stage, Overhung Model Test Pump

A model test bench with a single stage overhung pump, to compare “old” vs “new” impeller performances has been set up

Instrumentation prototype layout

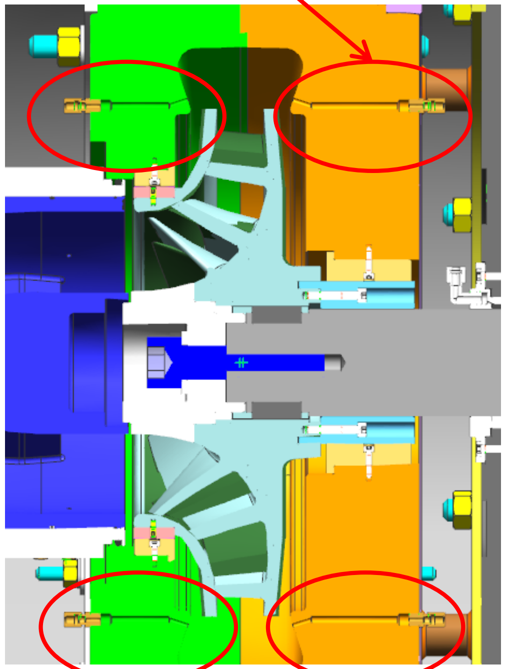
SEC 00: STAGE INLET

N°4 pressure taps eq. in 90° for static pressure analysis



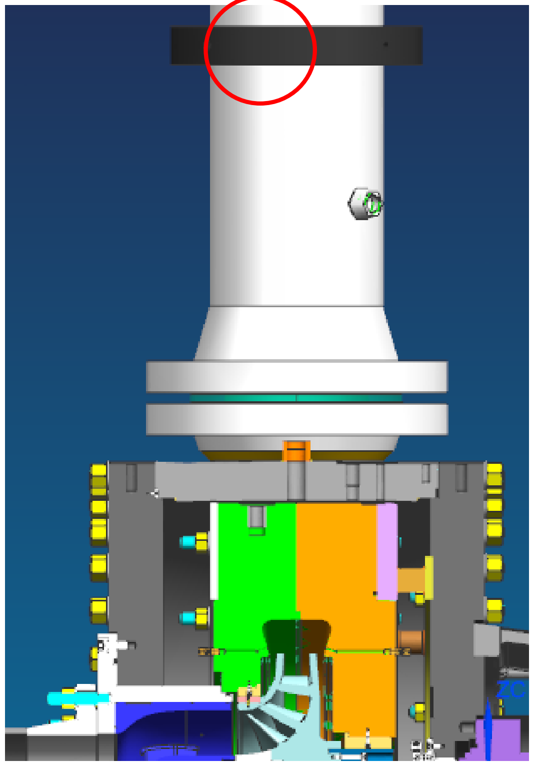
SEC. 19: IMPELLER OUTLET

N°4 pressure taps eq. in 180° for dynamic pressure analysis



SEC. 60: STAGE OUTLET

N°4 pressure taps eq. in 90° for static pressure analysis



PRESSURE SENSORS FOR PERFORMANCE/NPSH MEASUREMENTS

Test Matrix and Model Test Results

N. Test	Test type	n [rpm]	D_2[mm]	Notes
7 BLADES IMPELLER MAX DIAMETER				
1	BEP SEARCH	1700	355,6	Performance curve for BEP position
2	Performance	1700	355,6	Performance curve
3	NPSH 3%	1700	355,6	3% drop NPSHr curve
6 BLADES IMPELLER MAX DIAMETER				
4	BEP SEARCH	1700	355,5	Performance curve for BEP position
5	Performance	1700	355,5	Performance curve
6	NPSH 3%	1700	355,5	3% drop NPSHr curve
7 BLADES IMPELLER RATED DIAMETER				
7	BEP SEARCH	1700	343	Performance curve for BEP position
8	Performance	1700	343	Performance curve

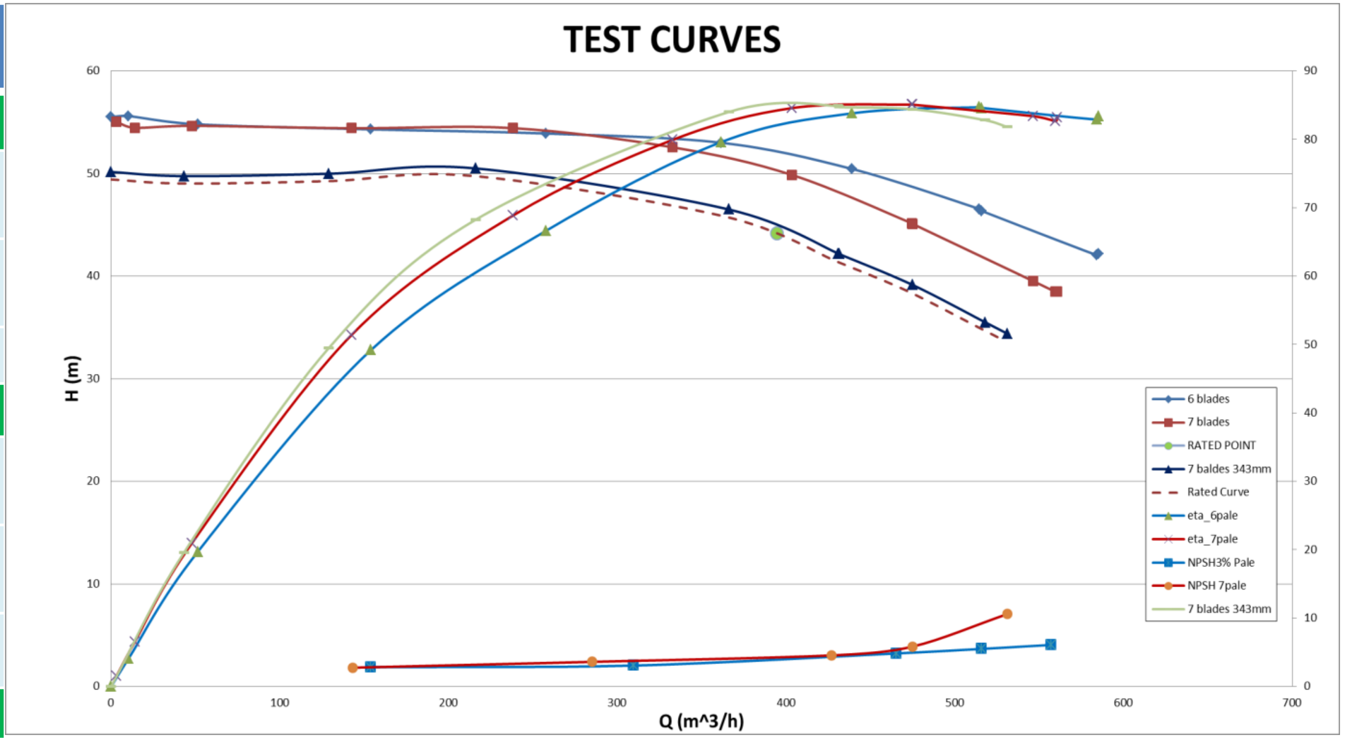
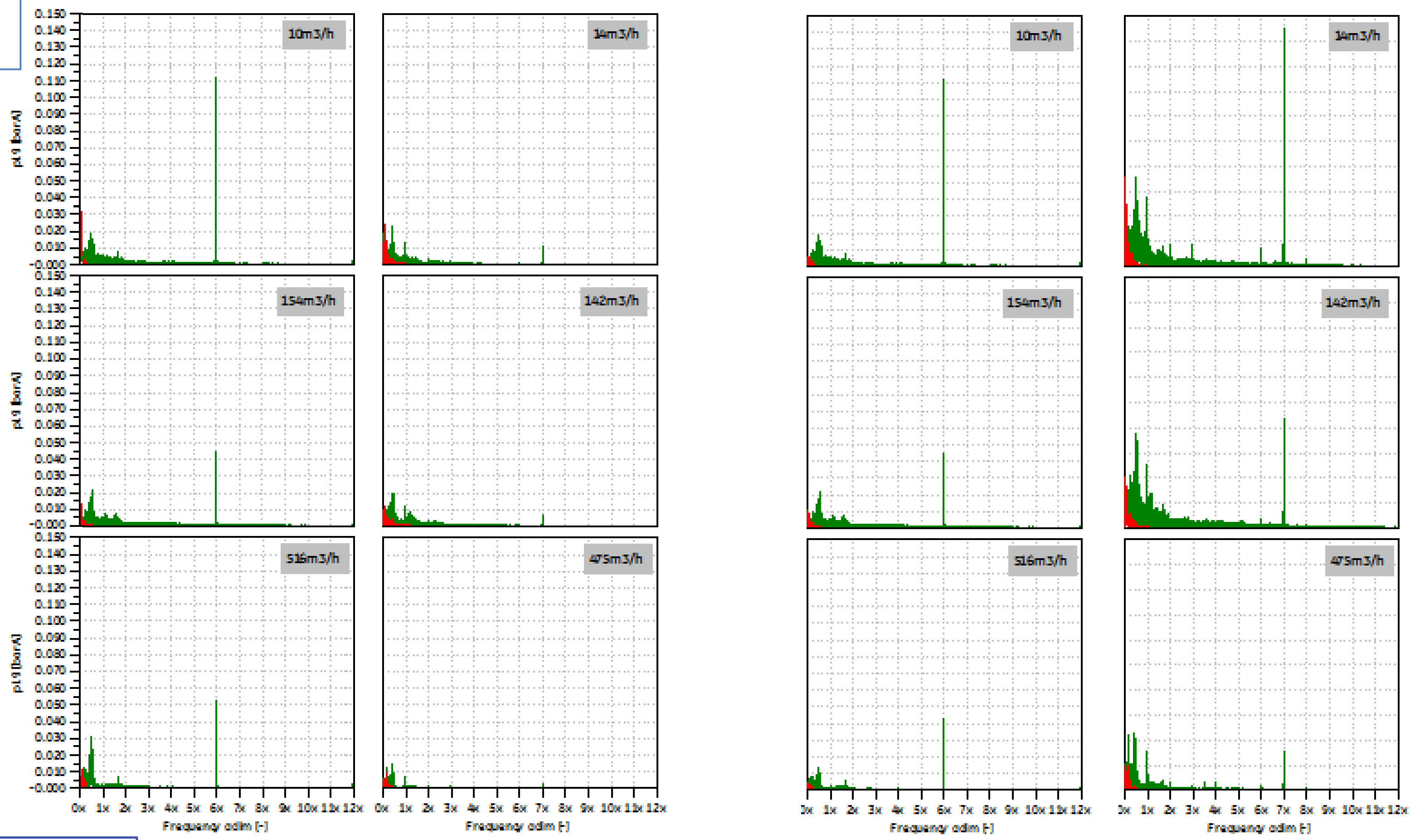


Figure 20 – Measured Performances Aligned With Expected Curves

Dynamic Pressure Analysis on SEC. 19 -Impeller Outlet

Back Signal | **Front Signal**

FLOW RATE & VIBRATION REDUCTION %	6 blades impeller (6x)	7 blades impeller (7x)	6 blades impeller (6x)	7 blades impeller (7x)
Q~10m3/h				
Q=30%Qbep				
Q=Qbep				



Dynamic pressure signal measured:

- Impeller back reduced at 1/5
- Impeller front comparable

Site Vibration Feedback after New Impeller Installation

Amine Pump Healthiness	RUN	SEAL POT		VIBRATION					
		DE	NDE	NDE-X	NDE-Y	DE-X	DE-Y	AXIAL-1	AXIAL-2
	Limit	LL: 20 HH: 100	LL: 20 HH: 100	H: 60 HH: 70	H: 60 HH: 70	H: 60 HH: 70	H: 60 HH: 70	H: 0.6 HH: 0.8	H: 0.6 HH: 0.8
P5-0201A	-17	8	-1	0.0	0.0	0.0	0.0	1.0	0.2
P5-0201B	4432	70	67	46.0	26.5	86.0	88.9	0.3	0.3
P5-0201C	0	61	57	1.9	1.8	0.6	0.6	0.1	0.1
P5-0201D	4574	59	61	27.5	15.6	32.8	37.8	0.5	0.4
P5-0201E	957	64	63	62.4	0.0	61.8	79.0	0.3	0.4
P6-0201A	4442	64	67	21.3	20.0	42.2	34.8	0.3	0.2
P6-0201B	4498	75	68	92.2	40.0	26.0	25.0	0.7	0.8
P6-0201C	-7	62	64	0.8	0.8	0.6	0.7	0.4	0.2
P6-0201D	-10	15	-1	0.0	0.0	0.0	0.0	1.0	1.0
P6-0201E	1101	55	60	28.5	28.8	21.0	15.8	0.4	0.4

Figure 21 – No Contact Pump Vibration with 6 Vane Impeller

	RUN		PUMP VIBRATION					
	Pump	HPRT	NDE-X	NDE-Y	DE-X	DE-Y	AXIAL-1	AXIAL-2
			um	um	um	um	mm	mm
	Limit		H: 60 HH: 70	H: 60 HH: 70	H: 60 HH: 70	H: 60 HH: 70	H: 0.6 HH: 0.8	H: 0.6 HH: 0.8
P5-0201A	4552	0	11.6	12.6	21.1	16.7	0.3	0.2
P5-0201B	-2	0	2.2	1.5	1.4	1.0	0.3	0.3
P5-0201C	4498	4496	14.8	25.1	7.4	12.6	0.1	0.1
P5-0201D	-5	-12	0.8	0.7	0.6	0.6	0.2	0.2
P5-0201E	903		13.4	14.3	16.7	18.2	0.4	0.4
P6-0201A	4515	4513	25.8	29.6	37.1	28.1	0.1	0.1
P6-0201B	4513	0	30.1	24.4	13.7	17.2	0.4	0.3
P6-0201C	4515	-15	25.9	24.7	20.7	25.6	0.1	0.2
P6-0201D	-7	0	0.8	0.8	0.8	0.7	0.2	0.3
P6-0201E	-4		0.6	0.6	0.6	0.6	0.3	0.3

Figure 22 – No Contact Pump Vibration with 7 Vane Impeller

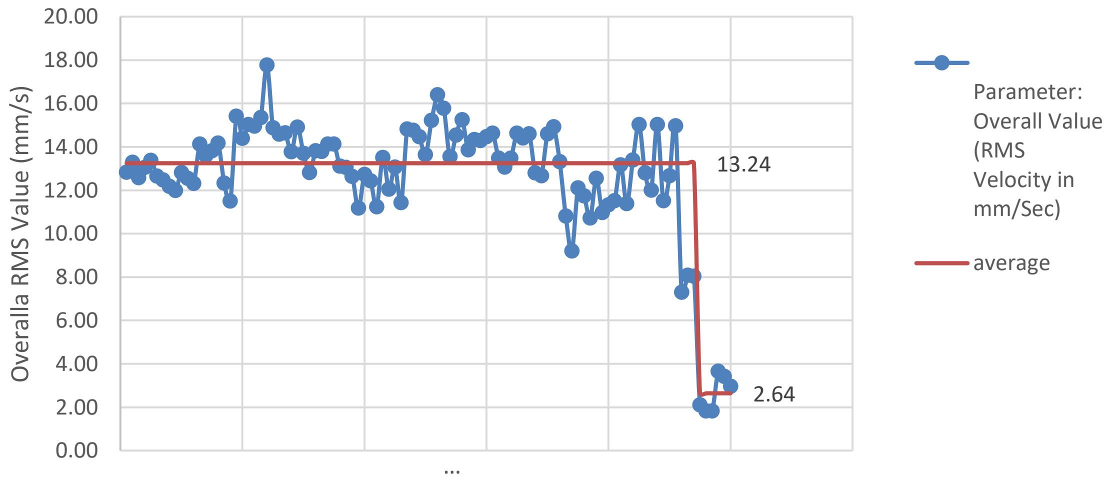


Figure 23 - Pump Hor Vel Overall Value

Thanks to the introduction of the new impeller Vibration have been reduced up to 75% in no contact value and 80% in velocity value

CONCLUSION

NEW IMPELLER HAS GUARANTEED:

- HYDRAULIC PERFORMANCES AND NPSHr WITH NEW 7 BLADES IMPELLER ALIGNED WITH EXPECTED CURVES
- DESING POINT CLOSER TO OPERATING POINTS (81% -105% $Q_{DESIGN\ NEW}$ VS 64% - 83%$Q_{DESIGN\ OLD}$)
- REDUCTION OF FLUID DYNAMIC FORCES
- NEW IMPELLER HAVE IMPROVED RELIABILITY OF THE PUMP REDUCING SENSIBLY VIBRATION

LESSON LEARNED

- High energy density pump and high suction specific speed shall not operate @ partial loads (Pump was operating at 64% of Qbep and 83% of Qbep), since operating statistics hinted to an accumulation of cavitation and vibration damage.

- During design phase, it’s important to verify the margin between VPF (Vane passing frequency) and Bearing Resonance frequency, to avoid high vibration during normal operating condition.

- Model test, and consequently CFD validation, gives us the chance to predict pump behavior at off design condition highlighting pressure pulsation entity at impeller exit and than the alternating forces acting on statoric parts.

- Strong connection with the Customer lead to design the hydraulic parts that match service performance expectation